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# INTEGRATING ENVIRONMENTAL CONTROL SYSTEM FUNCTIONALITY INTO A SCHEME OF THERMAL MANAGEMENT SYSTEM EVALUATION METRICS FOR MILITARY VEHICLES

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**ABSTRACT**

*Building upon the foundation of thermal management system metrics developed and published from earlier studies, this paper addresses the incorporation of the vehicle Environmental Control Systems (ECS) into this framework. Case studies are presented that look at the implications of several ECS alternatives on a conceptual military, hybrid electric vehicle platform through exemplar calculations of thermal management system (TMS) metrics. Utilization of these metrics allows for the comparison of design alternatives through a conceptual design case study. Two sets of case studies are evaluated in tandem within this study. In the first, the effect of cabin air-cooled component heat rejection is compared to direct TMS liquid-cooled heat rejection alternatives. The considerations of performance, packaging and reliability concerns are discussed. In the second set of case studies, variations in vapor compression cycle thermodynamic states are considered to provide metric-based guidance on design alternatives.*

## INTRODUCTION

Today's advanced military platforms can generate, convert, store and utilize power in a variety of ways. The functions of control, management and rejection of waste heat are a consistent challenge particularly in adverse environmental conditions. Vehicle thermal management systems (TMS) must manage and ultimately reject this waste heat to maintain reliable equipment operation. Beyond equipment survivability concerns, crew comfort has proven critical to mission performance and endurance. Further, as the assemblage of vehicle electronics continues to grow and more components are integrated that reject waste heat to vehicle cabin air, the importance of the vehicle environmental control system (ECS) as an integral subsystem of the overall vehicle TMS grows. Waste heat rejected to cabin air from electrical components is typically rejected through the ECS condenser that is often an integral component of the vehicle TMS heat exchanger stack.

In an attempt to provide direct measures of ground vehicle thermal management system design, several key metrics have been identified [1-2] that provide a measure of the performance and packaging based effectiveness. To date, those studies have neglected this inclusion of the ECS into this schema. This study looks to integrate the ECS functionality into the TMS metrics framework through an exemplar design study on a conceptual hybrid electric vehicle platform. Further, platform design alternatives associated with the ECS are evaluated within the overall metrics framework.

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Environmental control systems are becoming an integral part of a vehicle thermal management system. This is particularly true for under-armor applications where maintaining internal cabin temperature is crucial for both crew comfort and component performance. As today's military platforms mature, there is an ever increasing demand for more on-board power consumption. Many of these components are located within the vehicle cabin enclosure and often utilize or rely upon the cabin air to maintain thermal operational margin.

Internal component placement within the vehicle, specific heat rejection needs and component design particulars may impose or dictate a preferred cooling strategy for a given component. However, generally there are two choices available to the design engineer: cooling with cabin air or utilizing the on-board liquid cooling afforded by the vehicle TMS. These design considerations impact the design of the individual component, the packaging and routing within the vehicle structure and performance of the entire vehicle platform. Variations in the cooling strategy of on-board components (cabin air versus liquid-cooling) are investigated in this study.

Traditionally, environmental control systems have often been treated separately from the overall vehicle thermal management system; considered as an auxiliary automotive subsystem rather than integral to operation of the overall TMS. However, as the thermal burden on the ECS continues to grow, this distinction is no longer viable. In a typical application heat rejection from the ECS condenser utilizes the primary vehicle cooling air pathway. The thermal burden imposed on vehicle cooling air and the performance impact on subsequent TMS heat exchangers must be considered.

Although the process of evaluating TMS metrics could be applied to any vehicle topology, this study focuses upon a particular conceptual platform. More specifically, this study uses a conceptual, full-series hybrid platform as a case study exemplar from a design consideration standpoint. The process of TMS metrics evaluation could equally apply to an existing platform as well. The analysis requires a ground-up approach that considers vehicle operating requirements, individual component performance models and ambient conditions.

Typical environmental control systems utilize a vapor-compression cycle to remove waste heat from the cabin air and reject it to the environment. For this study, an idealized vapor-compression cycle was used to model the performance of the ECS. The effects of the sub-ambient cooling of the ECS are investigated in a comprehensive vehicle TMS analysis. Further, several thermodynamic operating states, corresponding to condenser and evaporator temperatures, are illustrated to demonstrate the impact on overall system performance.

## Thermal Management System Metrics

The thermal management system metrics included in this study follow those of previous studies [1-2]. Two primary classes of metrics have been identified: performance-based metrics and packaging-based metrics. Performance-based metrics characterize the functionality of the vehicle thermal management system against the vehicle's tractive power. These measures have been identified as:

$$\begin{matrix} \text{Thermal Load} \\ \text{Metric} \end{matrix} = \frac{\text{Vehicle Thermal Load}}{\text{Tractive Power}} \tag{1}$$

$$\begin{matrix} \text{Thermal Hotel} \\ \text{Load Metric} \end{matrix} = \frac{\text{Thermal Hotel Load}}{\text{Tractive Power}} \tag{2}$$

$$\begin{matrix} \text{Thermal Rejection} \\ \text{Effectiveness} \end{matrix} = \frac{\text{Vehicle Thermal Load}}{\text{Thermal Hotel Load}} \tag{3}$$

Packaging-based metrics characterize the real estate usage and weight allocations of the overall thermal management system against the overall vehicle and have been defined as:

$$\begin{matrix} \text{TMS Volume} \\ \text{Metric} \end{matrix} = \frac{\text{TMS Volume}}{\text{Vehicle Volume}} \tag{4}$$

$$\begin{matrix} \text{TMS Weight} \\ \text{Metric} \end{matrix} = \frac{\text{TMS Weight}}{\text{Vehicle Weight}} \tag{5}$$

## VEHICLE SYSTEM OVERVIEW

The vehicle used in this study is a conceptual 35 ton, tracked, full-series hybrid. A general depiction of the electrical architecture is shown in Figure 1. Mechanical power generated by an internal combustion prime power unit (PPU) is converted to distributed DC power through an integrated starter generator (ISG) and AC/DC power converter. The DC power generated is utilized throughout the vehicle platform to drive electrical traction motors for mobility, pump and fan drives for thermal management system operation, ECS compressor and evaporator fans (not shown), on-board mission electronics loads, export power functions and stored or retrieved through an on-board energy storage system (ESS) battery.

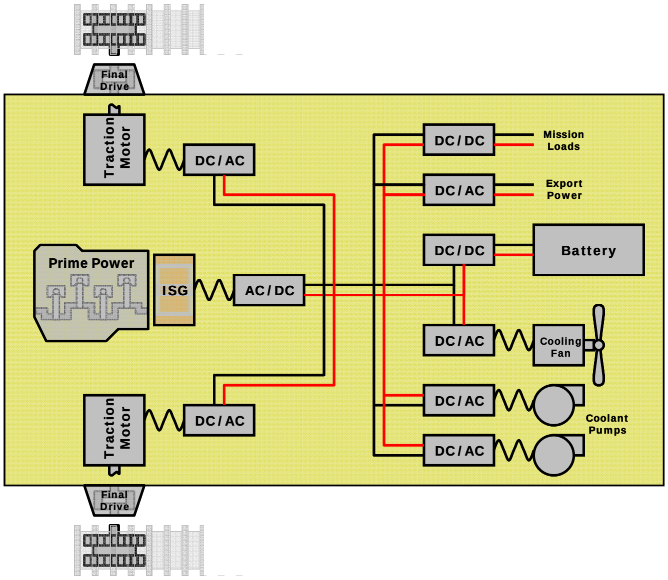


**Figure 1:** Overall Vehicle Electrical Architecture

Performance models and/or efficiency measures, where appropriate, have been assumed for each of the components on the platform to generate loading conditions, energy balances and heat rejection requirements. The specific case study in question is a vehicle top speed requirement of 65 kph on a 49°C on a hot, dry day (Category A1 [3]). This steady-state operational condition precludes the inclusion of the ESS or export power functionality of the platform. The top speed condition is a typical stressing scenario to the vehicle thermal management. Although high tractive efforts conditions are also a typical TMS stressing design condition, this study focuses upon the top speed scenario.

The thermal management system used in this study is shown in Figure 2. The TMS primary cooling fan draws air through the cooling stack air pathway removing waste heat from the ECS system, low temperature (LT) cooling system and high temperature (HT) cooling system in succession. Ballistic grilles at both the inlet and outlet of the cooling pathway have been included to represent pressure losses and estimate fan power consumption.

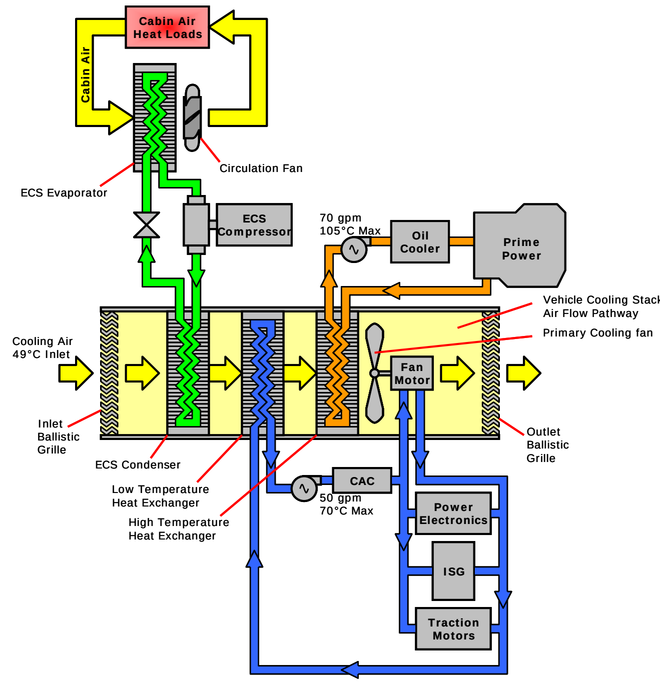


**Figure 2:** Overall Layout of the Vehicle Thermal Management System

The ECS system is a standard vapor compression cycle consisting of a compressor, evaporator, expansion valve and condenser unit. Cabin air is recirculated across the evaporator unit to accept waste heat from ambient, solar, personnel, fresh air make-up and air-cooled mission electronics components.

The LT system is assumed to require 50 gpm of 50/50 ethylene-glycol mixture (EGW) with a maximum allowable return temperature of 70°C. This system is assumed to provide cooling for the PPU charge air cooler (CAC), fan motor, power electronics units, ISG and traction motors. The HT system is assumed to require 70 gpm of 50/50 EGW at a maximum allowable return temperature of 105°C. This system provides cooling to the PPU oil cooler and engine block.

## SIMULATION METHODOLOGY

Heat loads included in this study were derived through an iterative energy balance method. The disposition of these heat loads are shown graphically in Figure 3. Estimated heat loads were either derived from assumed conditions (such as ambient and personnel loads) or calculated as part of a larger vehicle energy balance iterative procedure.

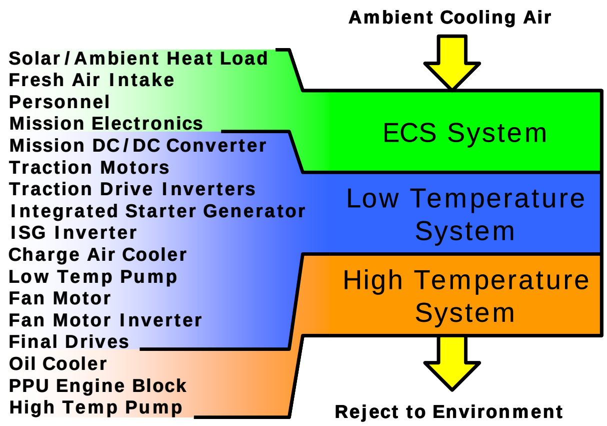


**Figure 3:** Disposition of Heat Sources to TMS Subsystems

The process begins with the mobility requirements associated with pushing a 35 ton vehicle at 65 kph along a flat surface (assumed parameters shown in Table 1). These parameters were used to determine the vehicle tractive power requirement (202.2 kW or 101.1 kW per side).

| Mobility Parameters | Value | Units |
|---|---|---|
| Vehicle Weight | 35 | tons |
| Top Speed Requirement | 65 | kph |
| Frontal Area | 6 | $m^2$ |
| Drag Coefficient | 1 | - |
| Rolling Resistance | 65 | lb / ton |
| Slope | 0 | % |
| Sprocket Radius | 0.3 | m |
| Final Drive Ratio | 12.5 | - |

**Table 1:** General Vehicle Mobility Parameters

Assumed component efficiencies (shown in Table 2) were then used to determine the net power consumption from the vehicle DC bus. Estimated power consumption of the primary cooling fan and knowledge of the ECS compressor power (derived from knowledge of the cycle Coefficient of Performance and net heat load) allowed for determination of the required power production from the ISG to provide steady-state operating conditions.

| Efficiency Parameters | Efficiency (%) |
|---|---|
| Final Drives | 96 |
| Traction Motors | 92 |
| Traction Motor Inverters | 95 |
| DC / DC Converter (Mission Power Needs) | 95 |
| Pump Drives | 97 |
| Fan Motor | 93 |
| Fan Motor Inverter | 97 |
| Generator | 93 |
| Generator Inverter | 97 |

**Table 2:** Component Efficiency Parameters

An assumed engine map (shown in Figure 4) allowed for determination of the PPU engine speed (rpm), torque production and specific fuel consumption. Knowledge of the engine fuel consumption rate allowed for estimation of waste heat generation for the engine exhaust, CAC, oil cooler and primary engine block coolant through assumed ratios of surplus fuel power (exhaust 55%, CAC 14.4%, oil 15.3% and coolant 15.3%).

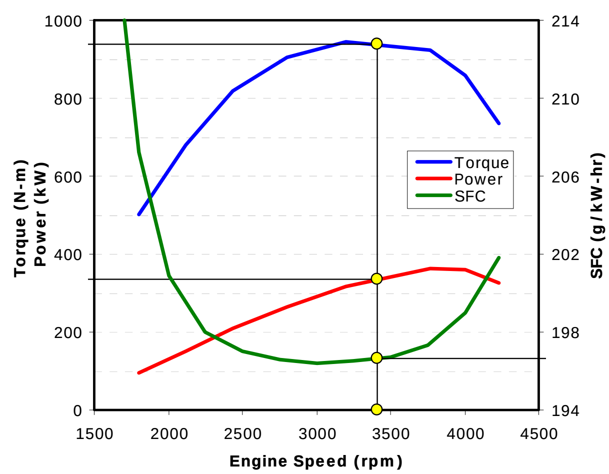


**Figure 4:** Prime Power Unit Torque, Speed and Fuel Consumption Map

Heat loads to each of the three systems (ECS, LT and HT) were then used to solve for the required heat exchanger depths assuming an inlet frontal area of 0.8 × 1.0 m. The heat exchanger characteristics and coefficients of the performance models used are shown in Table 3 and Table 4.

| HEX Characteristics | Value | Units |
|---|---|---|
| Flow area / frontal area ratio | 0.780 | - |
| Air-side hydraulic diameter | $3.51 \times 10^{-3}$ | m |
| Specific surface area | 886 | $m^2/m^3$ |
| HEX frontal area | 0.80 | $m^2$ |

**Table 3:** Heat Exchanger Surface Characteristics

The performance models for the heat exchanger cores used included a heat transfer correlation given by:

$$St\,Pr^{2/3} = a_0 + a_1 Re^{-1} + a_2 Re^{-0.5} + a_3 Re^{-0.2} \quad (6)$$

and a friction factor (pressure loss) correlation given by:

$$f = b_0 + b_1 Re^{-1} + b_2 Re^{-0.5} + b_3 Re^{-0.2} \quad (7)$$

The coefficients for the heat exchanger models are given in Table 4. Heat exchanger analysis followed well established practices [3] to determine each heat exchanger depth in a stepwise fashion. In other words, the inlet air conditions to the LT heat exchanger are dependent upon heat rejection from the ECS condenser. Similarly, the inlet air to the HT heat exchanger is dependent upon total heat rejection from the ECS condenser and LT heat exchanger.

| HEX Performance Correlations | Constant | Value |
|---|---|---|
| Stanton Number Correlation $StPr^{2/3}$ | $a_0$ | 0.0062 |
| | $a_1$ | 1.3657 |
| | $a_2$ | 0.3054 |
| | $a_3$ | -0.0291 |
| Friction Factor Correlation $f$ | $b_0$ | 0.0380 |
| | $b_1$ | 1.6011 |
| | $b_2$ | 1.8916 |
| | $b_3$ | -0.2401 |

**Table 4:** Coefficients of the Heat Exchanger Performance Model

An inlet grille model was developed that has similar pressure-flow characteristics to existing military variants. Total pressure loss was then tallied to give an estimate of fluid power imposed by the cooling fan.

An assumed fan efficiency and performance curve, shown in Figure 5, allowed for an updated estimate of the fan power consumption. It should be noted that each of the principle simulations were performed using 10 kg/s (@ 49°C) inlet air to the cooling stack.

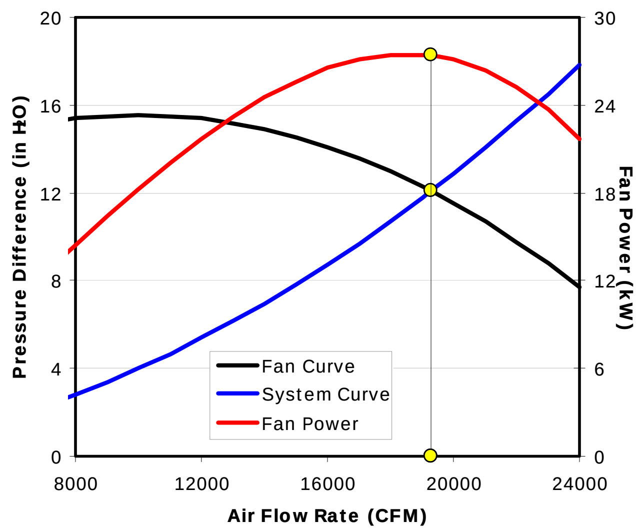


**Figure 5:** Fan Pressure, Fan Power and System Curves

Updated fan power numbers results in the need to iterate upon the total system energy balance. This re-establishes a specific demand from the PPU, an updated version of the engine heat loads and a new solution of heat exchanger depth and fan power estimates. This process typically converges within a couple of iterations with internal iterative calculations adjusting for air and fluid property temperature dependencies.

## SIMULATION RESULTS

The baseline case assumed 15 kW of mission electronics load. Further, it was assumed that 100% of that power is rejected as waste heat to the cabin air. The nomenclature used throughout is mission heat load fraction to cabin air. For the baseline case, this fraction is unity. Subsequent simulations were performed for heat fractions of 0.75, 0.50, 0.25 and 0 as well. The remaining mission heat load was transferred to the LT cooling system. In addition, for the baseline case, the ECS vapor-compression cycle limits were assumed to operate between 60°C for the condenser and 20°C for the compressor.

The overall vehicle energy balance is shown in Figure 6. Fuel power, determined from fuel consumption rate and PPU power production results in thee tractive power and fan power to accelerate the cooling air stream. Waste heat from this process, including additional ECS loads (ambient, personnel and fresh air intake) is manifest as waste heat through the exhaust and thermal management system heat rejection.

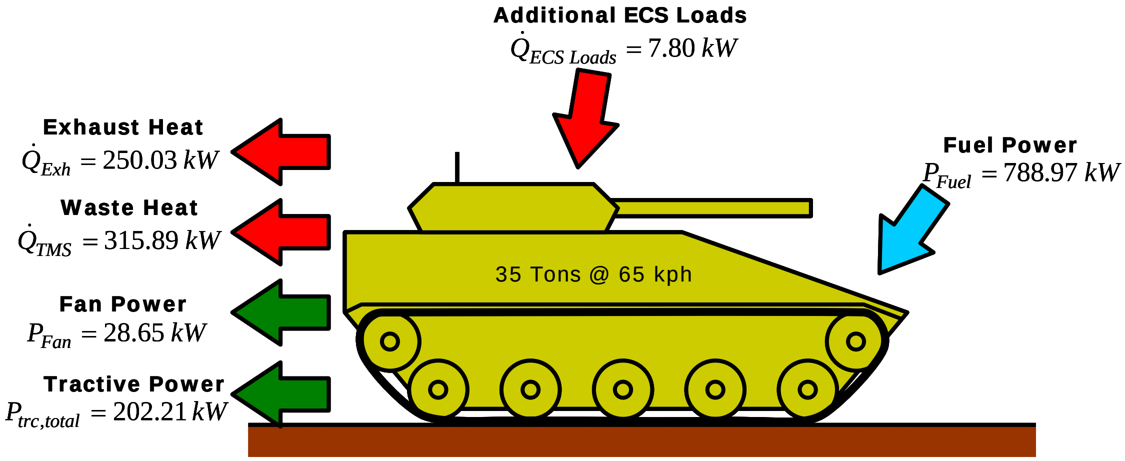


**Figure 6:** Overall Vehicle Energy Balance

To understand the simulation results that follow and subsequent metrics evaluations, it can be informative to understand the representative temperatures at critical locations in the heat exchanger stack. An example case, for a 60°C ECS condenser simulation is shown in Figure 7. Cooling air flow (shown in yellow) traverses from left to right and the individual utility streams (ECS refrigerant, LT coolant and HT coolant) pass from right to left. The reader should note how the cooling air temperature increases as it passes from one heat exchanger core to the next. Heat exchanger core sizes are largely driven by inlet temperature differences between the coolant and air streams.

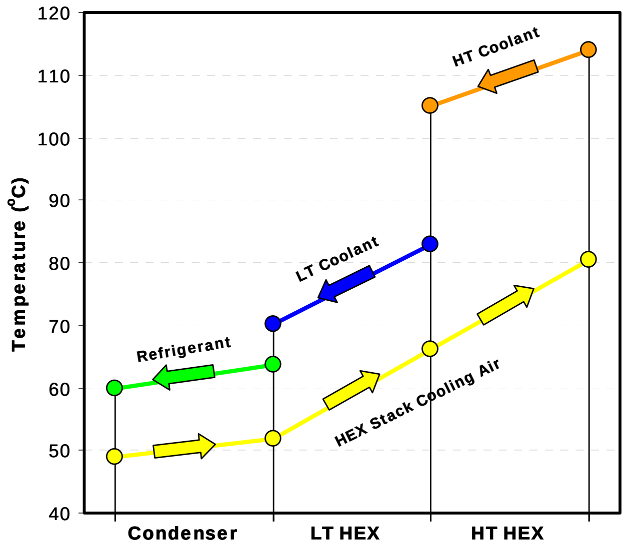


**Figure 7:** Heat Exchanger Stack Temperature Trends for 60°C Condenser Temperature Cases

The baseline solution heat exchanger core thermal loads and calculated core depths are shown in Table 5.

| Baseline Case HEX Loads | Value (kW) | Depth (m) |
|---|---|---|
| ECS Condenser Core | 28.2 | 0.049 |
| Low Temperature HEX Core | 144.7 | 0.230 |
| High Temperature HEX Core | 143.0 | 0.111 |
| Totals | 315.9 | 0.389 |

**Table 5:** Baseline Case Heat Exchanger Loads and Calculated Heat Exchanger Depth

Subsequent solutions with differing mission heat load fractions and vapor-compression cycle temperature limitations each generate a unique set of heat exchanger thicknesses. The four ECS vapor-compression cycles utilized in this study are shown in Figure 8. The simulations use idealized vapor-compression cycles. A more sophisticated study would include non-ideal cycle operations. Simulations have been performed for condenser temperatures of 60 and 70°C and evaporator temperatures of 20 and 10°C.

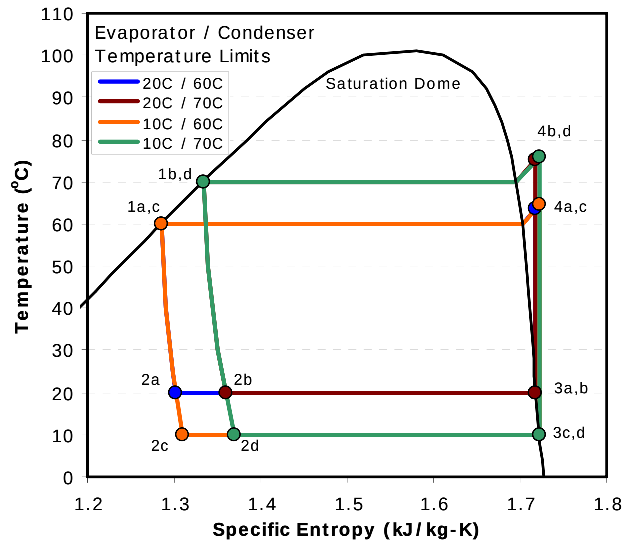


**Figure 8:** T-S Diagrams for Idealized R134a Vapor Compression Cycles

At higher condenser temperatures, the temperature difference between the condenser and cooling air is higher as shown in Figure 9. As expected, this results in a net smaller condenser core depth. However, a greater temperature difference between the evaporator and condenser temperatures implies a lower vapor-compression cycle Coefficient of Performance (COP). Lower COP implies a greater quantity of input work (compressor power) to transfer a given amount of heat. Greater power draw and losses through compressor inefficiency imposes a greater heat rejection demand.

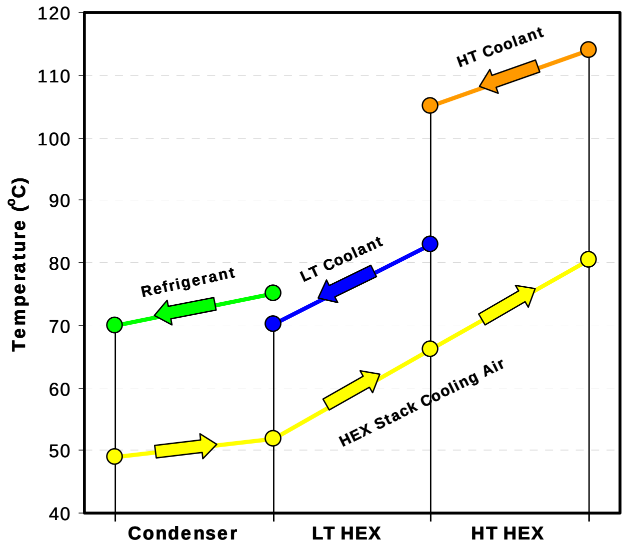


**Figure 9:** Heat Exchanger Stack Temperature Trends for 70°C Condenser Temperature Cases

The resulting calculated heat exchanger depths are shown in Figure 10 as a function of mission heat fraction to cabin air for each of the four vapor compression cycles. Interestingly, the minimum net heat exchanger core thicknesses (and, hence, smallest volume) were found with the 20°C evaporator and 70°C condenser temperatures. The reason for this is that the improvements found by increasing the heat rejection temperature in the primary cooling stack surpass the increased heat load penalties associated with the 'less efficient' cycle operation.

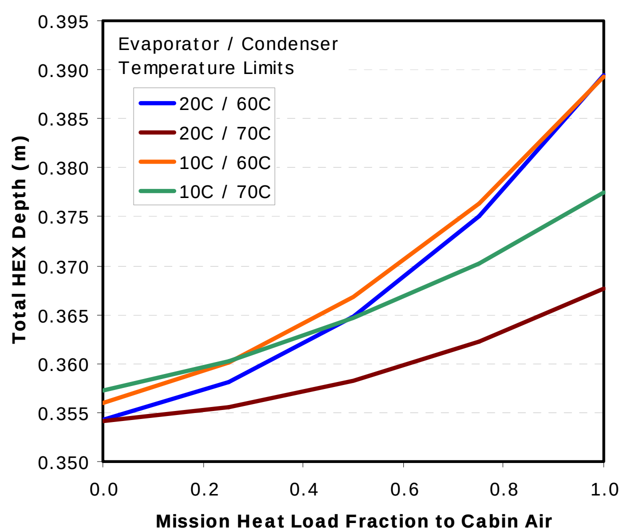


**Figure 10:** Total Heat Exchanger Depth

Although the greater condenser temperature reduces the net cooling stack heat exchanger volume, this effect is not mirrored by simultaneously decreasing the evaporator temperature. As seen by the 10°C evaporator cases, the net effect of a less efficient vapor-compression cycle becomes dominant over the baseline case of 20°C evaporator / 60°C condenser. However, it should be noted that this effect is only dominant at reduced net heat loads to the cabin air. As heat is transferred from the low temperature cooling system to the cabin air (higher mission heat load fraction), the reduced evaporator temperature cases tend to approach and surpass the baseline simulation. Further study is needed to determine if this effect is a direct function of the mission heat load fraction or the percentage heat load to the overall vehicle TMS. Also, the effects of non-idealized vapor-compression cycles need to be taken into consideration.

It should be emphasized that no provisions have been made to size the evaporator core(s) in this study. Multiple evaporators may be used in an under-armor vehicle to provide a zoned approach to cooling and packaging considerations and condensation effects may dictate the best orientation. It can be generally stated, however, that reduced evaporator temperatures will result in decreased evaporator core sizes and circulation fan requirements.

Before compiling the TMS metrics for these cases, another factor should be considered. All simulations shown in this study have used 10 kg/s of cooling air to provide a uniform comparative baseline. Figure 11 shows the total heat exchanger stack depth as a function of cooling air mass flow rate. A detailed design optimization needs to consider alternative flow rates as parameter.

As a general trend, the additional burden of increased sub-ambient cooling for increased mission load fraction is clearly observed. This is the product of both increased ECS demand and lower rejection temperature of the ECS condenser opposed to the LT system. The inflection point shown at roughly 11 kg/s in Figure 11 is the product of reaching the power inflection point in the engine performance map (see Figure 4). Proper matching of the resultant pressure losses and fan performance curves need to be matched with overall system power production and usage to ultimately yield an optimal configuration for a particular installation.

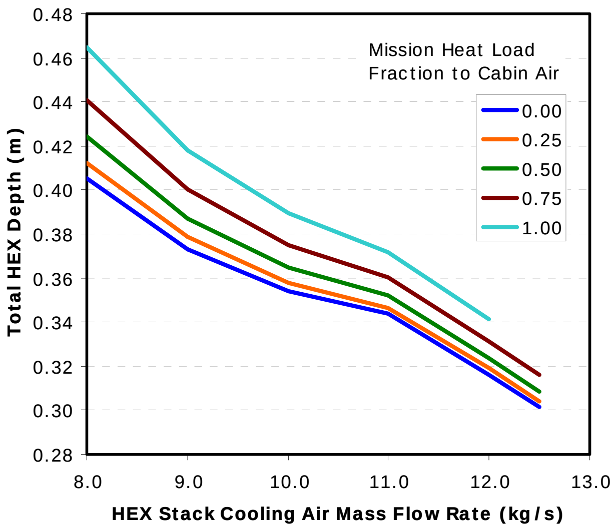


**Figure 11: Total Heat Exchanger Depth as a Function of Cooling Air Mass Flow Rate**

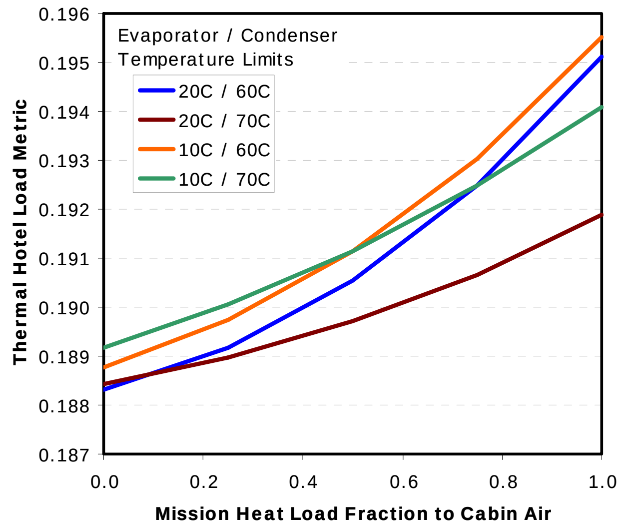


**Figure 12:** Thermal Hotel Load Metrics

## TMS METRICS

The thermal management system metrics for these case studies were calculated according to the equations (1)-(5). The first of these is the Thermal Hotel Load Metric as shown in Figure 12. Thermal hotel loads encompass the power consumption associated with pumps, cooling fans and the ECS compressor. No provisions for the evaporator/air circulation fans have been included. Also neglected were the power costs associated with additional LT fluid pumping and/or component cooling fans.

The Thermal Hotel Load Metric is scaled with respect to the vehicle tractive effort. The reasoning behind this scaling is to provide a measure that can be comparable across vehicle classes and weights. A lower hotel load implies that to meet the operational requirements, less power is consumed for internal management and heat dissipation.

The results of the Thermal Hotel Load Metric (Figure 12) follow the general trend of the total heat exchanger depths shown in Figure 10. Except at the lowest cabin air heat load, the 20°C evaporator / 70°C condenser case demonstrates the lowest overall metric. This plot demonstrates the importance of reducing the heat exchanger depth since fan power consumption is largely a result of heat exchanger pressure losses. The increased inefficiency of lower COP ECS cycles is surpasses by differences in fan power consumption.

The Vehicle Thermal Load Metric is defined as the ratio of the sum of thermal management system heat rejection (including ECS condenser load) to the vehicle tractive power. The Vehicle Thermal Load Metrics of these case studies are shown in Figure 13. Note that the net vehicle platform thermal losses are roughly 1.5 times the delivered tractive power. Even with the high efficiencies assumed for the electric machines and power converters, the steady-state operating conditions of the hybrid platform produce challenging TMS design needs.

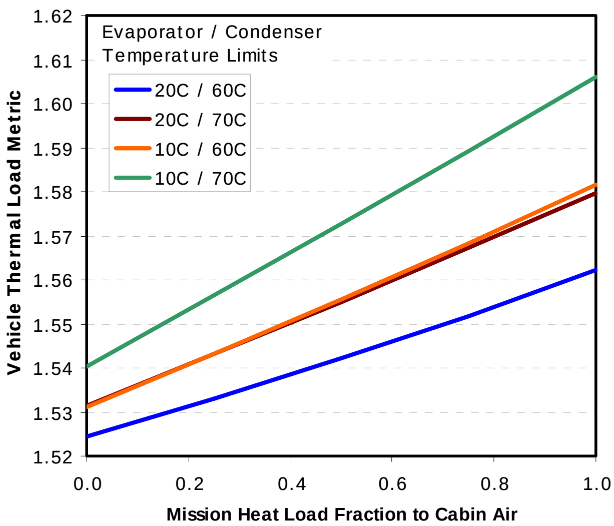


**Figure 13:** Vehicle Thermal Load Metrics

The Vehicle Thermal Load Metrics (Figure 13) demonstrate the effect of the efficiencies of the vapor-compression cycles. The highest COP vapor-compression cycle (20°C evaporator / 60°C condenser case) demonstrates the lowest overall system heat load. Alternatively, the lowest COP vapor-compression cycle (10°C evaporator / 70°C condenser case) demonstrates the highest overall system heat load.

Another representation of the previous two metrics can be found by the ratio between the two. This is referred to as the Thermal Rejection Effectiveness and is shown in Figure 14. This value is the ratio of the total vehicle thermal load to the vehicle thermal hotel load. Interestingly, the greater quantity of heat is rejected per unit of heat rejection power consumed for the less efficient cases.

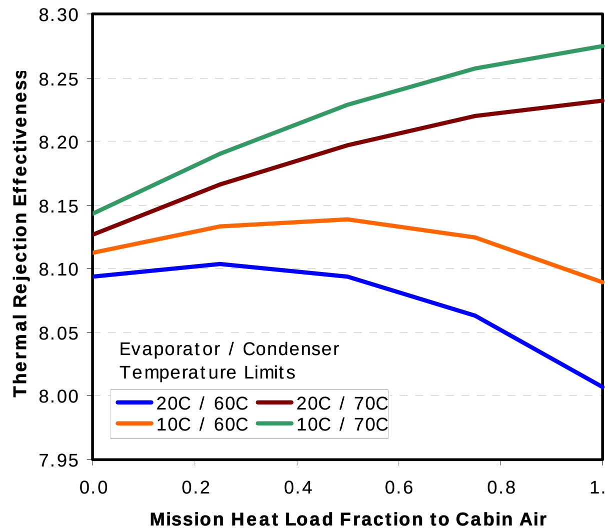


**Figure 14:** Thermal Rejection Effectiveness Metrics

These performance metrics alone do not dictate or provide a complete measure of optimal design. Although not included as part of this study, previous efforts [1-2] have also characterized an Operational Performance Margin Metric that quantifies the additional heat rejection margin of a given design.

The packaging-based metrics also play a considerable role in the comparison of one system or design alternative to another. Vehicle real estate is always at a premium and weight savings can be directly related to fuel consumption.

The vehicle TMS packaging-based metrics include the measures of vehicle weight and volume as shown in Figure 15 and Figure 16. Since no provisions for the packaging considerations of individual on-board component heat rejection (evaporators, fans, heat sinks, plumbing, etc.) have been considered in this study, the weight and volume measures are direct reflections of the heat exchanger size calculations. Assumed values for weights and volumes of individual components such as pumps, plumbing, coolant, fans, inlet/outlet grilles, ductwork were used to develop the numerical references shown in these figures. A detailed analysis would require significant auditing of the vehicle product structure, SWaP (size, weight and power) and mechanical layout to capture trapped, sway and maintenance access volumes.

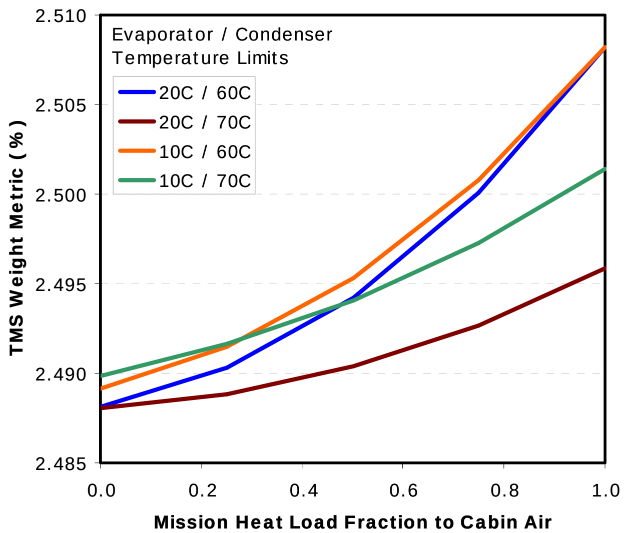


**Figure 15:** Thermal Management System Weight Metrics

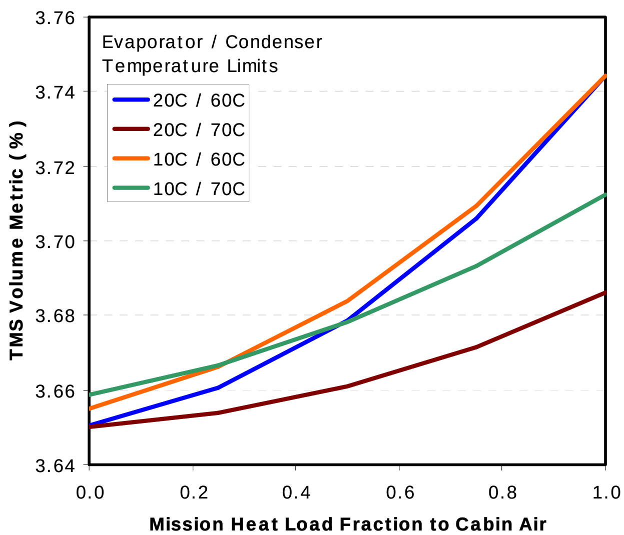


**Figure 16:** Thermal Management System Volume Metrics

## CONCLUSIONS

Vehicle environmental control systems need to be considered as integral to the overall vehicle thermal management systems. The heat rejection from ECS condenser units, typically upstream of vehicle cooling systems, impact the performance and design of downstream TMS heat exchanger cores and platform fan power consumption.

Trade-offs between liquid cooling and cabin air cooling of on-board electronics components further emphasize the need for ECS inclusion in TMS design evaluation. Liquid cooling of in-cabin power electronics shift the heat rejection from the ECS to the vehicle TMS. Analysis needs to consider both systems in an integrated manner. Packaging considerations will undoubtedly dominate decisions based upon component maturity, platform location, and component design particulars.

One general rule of thumb has been demonstrated in this study. Higher heat rejection temperatures lead to reduced heat exchanger sizes. For ECS systems, this implies that a higher condenser temperature results is preferable. However, this effect needs to be measured against the power consumption and system heat rejection requirements and ultimately, the packaging considerations to make the best design decision.

Higher efficiency ECS vapor-compression cycles do not necessarily promote the optimal solution. It is only through an overall thermal management system architecture audit and analysis process that conclusions can be drawn regarding the optimal system configuration.

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